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Recent advancements in liquid desiccant dehumidification technology

Yonggao Yin [a,b,*], Junfei Qian [a,b], Xiaosong Zhang [a,b]

[a] School of Energy and Environment, Southeast University, Nanjing 210096, China
[b] Ministry of Education of Key Laboratory of Energy Thermal Conversion and Control, Southeast University, Nanjing 210096, China

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ABSTRACT

Liquid desiccant dehumidification technology is becoming increasingly attractive due to its high efficient utilization of low-grade heat and its effectiveness in dehumidification. Using this technology, energy-efficient air conditioning systems have been developed, which demonstrated superiority over the traditional vapor compression type system by allowing both temperature and humidity to be controlled independently. This paper presented a state-of-the-art review of the research and development in this field, covering the topics of heat and mass transfer models, performance evaluation of liquid desiccant dehumidification and regeneration, and technology development of dehumidifiers and regenerators as the most important components of liquid desiccant systems. Meanwhile, many detailed systems using solar energy in desiccant cooling was reported, and some new applications of liquid desiccant dehumidification were also introduced.

Contents

1. Introduction

Increasing occupant comfort demand is leading to growing requirements for air conditioning, whereas the conventional vapor-compression refrigeration and air-conditioning systems are consuming high levels of electricity, and have dominated over 25% of total energy consumption in China. The CFCs/HCFCs refrigerants

* Corresponding author. Tel.: +86 25 83792722.
E-mail address: y.yin@seu.edu.cn (Y. Yin).

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Nomenclature

Cp	specific heat capacity (kJ/(kg °C))
c	salt concentration of desiccant solution (kg/kg)
D	diffusion coefficient (m^2/s)
d	air humidity ratio (g/kg)
H	height of the film falling (m)
h	enthalpy (J/kg)
Le	Lewis number
M_a	airflow (kg/s)
M_s	solution flow (kg/s)
NTU	number of transfer units
P	pressure (Pa)
r	the latent heat of vaporization of water (kJ/kg)
T	temperature (°C)
u	x-velocity component (m/s)
V	dynamic viscosity (m^2/s)

Greek symbols

ε	liquid desiccant dehumidification efficiency
μ	kinematic viscosity (Pa s)
τ	time (s)
α	thermal diffusivity (m^2/s)

Subscripts

in	inlet of labeled flow
out	outlet of labeled flow
a	air
L	liquid desiccant
equ	equilibrium
s	desiccant solution
sat	saturation

used in the conventional vapor-compression refrigeration, and air-conditioning systems can also bring some damage to the environment. It is now a priority to develop energy-efficient refrigeration and air conditioning technology.

Use of low-grade heat like solar energy for air conditioning is a suitable alternative. Liquid desiccant cooling systems have become increasingly attractive as compared conventional technologies due to many advantages, such as effective utilization of low-grade heat sources, and less damage to the environment. The liquid desiccant dehumidification technology has been used in industrial and agricultural industries, such as humidity control in textile mills, post harvests and low-temperature crop drying in stores. It is playing an increasingly prominent role in air-conditioning systems. Liquid desiccant-based air conditioning system can be driven by heat source below 80 °C for cooling and dehumidification, and then much electricity consumption can be avoided. Recently, liquid desiccant systems combined with vapor-compression chillers can develop into an energy-efficient independent temperature and humidity control for air conditioning system. The chiller could operate with high efficiency due to the lift of the evaporative temperature and reheating of the dehumidified air in traditional air conditioning is not necessary. Liquid desiccant dehumidification systems can dehumidify air by the direct contact between the air and concentrated solution, and achieve independent handling of sensible load and latent load of the processed air.

This paper presents a literature review of recent work concerning various aspects of liquid desiccant dehumidification technologies in an effort to improve the operating efficiency of its applications in the refrigeration and air conditioning systems.

2. Principle of liquid desiccant air conditioning

A liquid desiccant cooling system often consists of many different components, such as a dehumidifier, a regenerator, an evaporative cooler, heat exchangers, and so on. In the dehumidifier, the surface vapor pressure of the concentrated liquid desiccant with low temperature is lower than that of the processed air, and so the mass (water) transfer is from the processed air to the desiccant. After the dehumidification process, the desiccant solution is diluted and then pumped out to the regenerator where the surface vapor pressure of the diluted liquid desiccant with high temperature is higher than that of the ambient sweeping air. Thus, the mass (water) transfer is from desiccant to the processed air, and then the weak solution is concentrated. The surface vapor pressure difference between the liquid desiccant and air is the driving force for mass transfer. Thus, the diluted solution flowing out from the dehumidifier is preheated to increase its surface vapor pressure to improve the desiccant regeneration. The heat for the preheating process can be obtained by low-grade heat sources, such as solar energy, waste heat, and so on.

3. Heat and mass transfer model and performance evaluation

3.1. Modeling of the heat and mass transfer between air and desiccant

Very complex heat and mass transfer occurs in dehumidifiers and regenerators, which are very basic components in liquid desiccant air conditioning systems. Studying the performance of heat and mass transfer between liquid desiccant and air in the dehumidification/regeneration processes is important, as is promoting the industrialization and the application of the liquid desiccant air conditioning systems. Since Löf proposed the concept of the solar liquid desiccant air conditioning systems in 1955, many researchers have performed experimental and theoretical studies on the heat and mass transfer performance between liquid desiccant and air in liquid dehumidification/regeneration process, and have made a significant progress. Heat and mass transfer models are very basic and most essential for the performance evaluation of the dehumidifier and the regenerator. There are three types of common models: e.g., the finite differential diffusion equations model, the efficiency model, and the finite volume model.

3.1.1. The finite differential diffusion equations model

The finite differential diffusion equations model is used to theoretically study the heat and mass transfer law by solving the differential equation, which consists of a continuity equation, a momentum equation, a heat diffusion equation and a mass diffusion equation. They are based on conservation of mass and energy, and are combined with the theory of heat and mass transfer. Fig. 1 shows the schematic diagram of the liquid desiccant film falling on a vertical surface [1]. The liquid desiccant is distributed evenly on the vertical flat and is falling down by gravity. The moist air and the liquid desiccant flow through the channel by parallel-flow. First of all, the velocity distributions of the moist air and the liquid desiccant can be solved by the

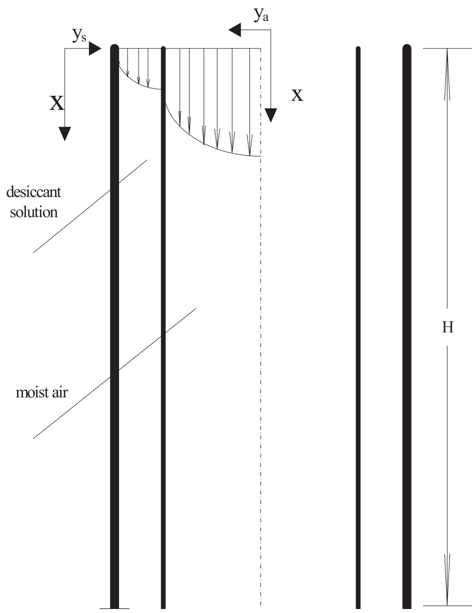

Fig. 1. Schematic of the liquid desiccant film falling on a vertical surface.

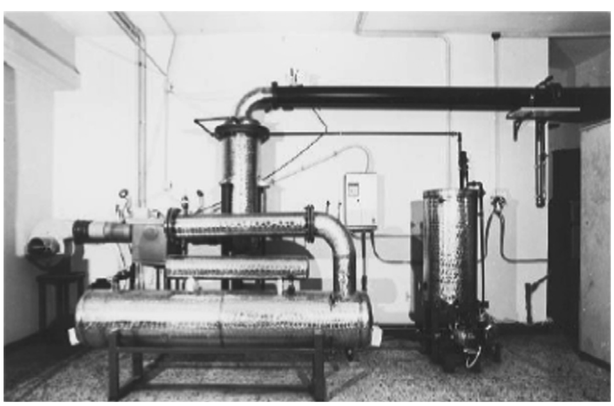

Fig. 2. Packed tower dehumidifying equipment.

Navier–Stokes momentum equation:

$$\rho \frac{D\vec{V}}{D\tau} = \rho \vec{f} - \nabla p + \mu \Delta \vec{V} \tag{1}$$

where $\vec{f}$ is the body force.

From the energy differential equations of the two-dimensional heat convection, the steady state convection–diffusion energy differential equations of the air and the liquid desiccant can be written as the follows:

$$u_a \frac{\partial T_a}{\partial x} = \alpha_a \frac{\partial^2 T_a}{\partial y^2} \tag{2}$$

$$u_s \frac{\partial T_s}{\partial x} = \alpha_s \frac{\partial^2 T_s}{\partial y^2} \tag{3}$$

Based on the conservation of the mass in the micro-control element, the two-dimensional steady state convection–diffusion mass controlling equations of the moist air and the liquid desiccant can be written as follows:

$$u_a \frac{\partial d}{\partial x} = D_a \frac{\partial^2 d}{\partial y^2} \tag{4}$$

$$u_s \frac{\partial c}{\partial x} = D_s \frac{\partial^2 c}{\partial y^2} \tag{5}$$

The mathematical model defined above was often used to study the developed and stable flow of the liquid desiccant film falling on a vertical or inclined flat surface, where the physical model for the heat and mass transfer would be simpler compared with packing towers. The velocity profiles of the air and desiccant could be determined based on the momentum equation. Usually it is necessary that some assumptions be made for further numerical solution of the diffusion equations. For example, the mass rate of the vapor transferred was lower compared to the film flow rate and did not destroy main flow of the liquid desiccant, and there was a thermal dynamic equilibrium in the gas–liquid surface between the air and desiccant. Therefore, the temperature, humidity and concentration profiles could be solved based on the finite differential diffusion equations model.

Many researchers carried out the theoretical investigation on liquid desiccant dehumidification and regeneration by adopting the finite differential diffusion equations model. Grossman [2–3] studied the liquid desiccant dehumidification process by adopting the control differential equations model in 1983, and obtained both the analytic solution and the finite difference numerical solution. The results showed that the mass transfer coefficient was dependent on *Re* and *Sc*, and the heat transfer coefficient was dependent on *Re* and *Pr*. Habib and Wood [4] adopted SIMPLER arithmetic to get the numerical solution for the laminar flow of the constant wall temperature falling film liquid dehumidification process. The thermal property and the solution falling film thickness changing with the falling film direction were taken into account. The result was in good agreement with the experimental data.

In 2004, Ali et al. [5] investigated the performance of the heat and mass transfer between the air and the falling solution film with a cross-flow configuration. The effects of addition of Cu-ultrafine particles in enhancing the heat and mass transfer process were also examined. It was found that low air Reynolds number and more Cu-volume fraction enhanced the dehumidification. Dai and Zhang [6] investigated the heat and mass transfer process in a cross-flow liquid desiccant dehumidifier packed by durable honeycomb paper. They compared the *Nu* at the liquid–air interface between the theoretical and numerical results. Zhang and Shi [7] determined the numerical solution of the laminar flow plate falling film liquid dehumidifier physical model under the convection boundary condition. It was presumed that the relationship between the solution concentration and the section temperature was linear. They obtained the influencing factors of both *Nu* and *Sh*. Yin et al. [1] demonstrated the numerical solution of the steady state adiabatic dehumidifier and internally cooled parallel flow plate falling film dehumidifier. The results showed that the solution temperature lift was the main reason for the dehumidification ability deduction. This model involved a complex calculation procedure and was used to analyze the heat and mass transfer of the regular laminar flow.

3.1.2. The empirical effectiveness model

The efficiency model mainly involved two efficiencies: humidity and enthalpy. According to the known inlet operating conditions of dehumidifier/regenerator, the outlet air parameters of the dehumidifier/regenerator can be solved if the humidity efficiency and the enthalpy efficiency were given. The humidity and enthalpy efficiencies were empirical correlations derived from many experimental data. As this model is relatively simple and easy for calculation, it is very popular for engineering and rough estimation.

In 1994, Chung [8] modified the humidity efficiency model proposed by Ullah (1988, ASME). Experiments on two liquid

desiccants and four bulk packing towers proved that the error of the efficiency model was less than 10%. The efficiency model was applied in some relative industry designs.

Martin and Goswami [9] studied the effects of the physical size on the heat and mass transfer based on the dimension analysis method. The humidity efficiency and enthalpy efficiency involved two liquid desiccants (LiCl and TEG) and three bulk packing towers, and were fitted against the previous experimental data. In dehumidification, the error of the humidity efficiency and enthalpy efficiency was less than 9% and 10% respectively, and the errors were less than 16% and 11% respectively in regeneration. Gandhidasan [10] simplified the dehumidifier into a black box to develop the humidity efficiency and the temperature efficiency model based on the conservations of mass and energy. This model agreed well with the experimental data of Fumo et al. [11] except that the temperature efficiency was negative under some experimental conditions. Abdul-Wahab [12] used triethylene glycol (TEG) as the desiccant to investigate the performance of the structured packing with three specific volumes (77, 100 and 200 m^2/m^3), and analyzed the effects of operating parameters on the moisture removal rate and dehumidifier effectiveness. On the basis of the experimental data of a cross-flow packed dehumidifier using LiBr–H_2O, Liu [13] developed empirical correlations of humidity and enthalpy efficiencies, which were determined by the enthalpy and humidity ratio difference, airflow rate and desiccant flow rate. Table 1 shows the efficiency correlations of the empirical model in the existing literature. The empirical effectiveness models were applicable under limited operation conditions, and it could be applied to the calculation only in the range of specific materials, certain configurations and under limited operation condition range.

3.1.3. *The finite control volume model*

The finite control volume model is most frequently used in the liquid dehumidification/regeneration. The basic idea is that it simplifies dehumidifier/regenerator to successive micro-control elements along the airflow or desiccant flow direction, and the heat and mass transfer occurs between the liquid phase and the gaseous phase in control volume where the liquid phase and the gaseous phase are uniform. In the model it is very critical to determine the heat and mass transfer coefficients between the liquid phase and the gaseous phase. Typically it is determined in two ways: (1) the double film model—the heat and mass transfer resistance of gas and liquid phases are considered; (2) the overall convective model—considering the overall convective heat and mass transfer resistance.

3.2. *The double film model*

The double film model suggested that the mass transfer resistance could be the sum of the resistances in liquid phase and gaseous phase, and simultaneously the mass concentration could be in equilibrium at the surface between the liquid and the gas. Onda et al. [15] proposed the formula of the moist surface-area coefficient for the mass transfer in bulk packed tower. The mass transfer coefficients of the distillation gas and liquid sides in the bulk packed tower were obtained based on the data in published papers. In 1985, Roberts et al. [16] evaluated three models: (1) the Sherwood-Holloway's model; (2) the Shulman's model; and (3) the Onda's model for air stripping of volatile organic contaminants in a countercurrent packed column. The results showed that the standard deviation of the wide working condition scope could be less than 21% in Onda's model. Öberg [17] investigated the heat and mass transfer between a liquid desiccant (triethylene glycol) and air in a packed bed absorption tower experimentally, and proposed the heat and mass transfer analogy method and the Ackerman correction coefficient, which used the mass transfer coefficients of the air–liquid side presented by Onda based on the micro-control element model. The results showed good agreement with the efficiency model presented by Chung [8]. Fumo et al. [11] studied the performance of a packed tower dehumidifier and a regenerator for an aqueous lithium chloride desiccant dehumidification system. The rates of dehumidification and regeneration as well as the effectiveness of the processes were assessed under different airflow rates, temperatures, humidity ratios and different desiccant flow rates, temperatures and concentrations. Al-Mutairi [18] and Al-Farayedhi [19] investigated the dehumidification model of the cross-corrugated and structured

Table 1
Efficiency correlations of the empirical model.

Application range	Correlation	Annotation
Random packing Countercurrent dehumidification LiCl and TEG	$\varepsilon_Y = \frac{1-\{0.25(G_{a,in}/G_{s,in})^{0.174}\exp\ 0.985(T_{a,in}/T_{s,in})]/(\alpha_t Z)^{0.184}\pi^{1.680}\}}{1-\{0.152\ \exp[-0.686(T_{a,in}/T_{s,in})]/\pi^{3.388}\}}$ where, $\pi = \frac{p_w(T_{s,in}) - p_s(T_{s,in},\xi_{in})}{p_w(T_{s,in})}$	Literature [8] Average error is $\pm 7\%$
Random packing Countercurrent dehumidification LiCl and TEG	$\varepsilon_Y = 1 - 48.345\left(\frac{M_s}{M_a}\right)^{(0.396(r_s/r_c)-1.573)}\left(\frac{h_{a,in}}{h_{s,in}}\right)^{-0.751}(a_w Z)^{(0.033(r_s/r_c)-0.906)}$	Literature [9] $3.5 < \frac{M_s}{M_a} < 15.4$
Structured packing TEG	$\varepsilon_Y = 0.061 + 0.25\ M_s - 0.00072\ a_w - 0.0107\ T_{a,in}$	Literature [12] $77 < a_w < 200$
Structured packing $550 \times 400 \times 350\ mm^3$ Cross-flow dehumidification LiBr	$\varepsilon_H = c_0(H_{a,in} - H_{equ,in})^{0.5641}(d_{a,in} - d_{equ,in})^{-0.6487}G_a^{-0.4435}G_s^{0.6201}$ $\varepsilon_Y = c_0 G_a^{-0.2804}G_s^{0.3657}$	Literature [13] $0.30 < G_{s,in} < 0.64$, kg/s $20.1 < T_{s,in} < 29.5\ ^\circ C$ $0.426 < X_{s,in} < 0.548$ $0.31 < G_a < 0.47$, kg/s $24.7 < T_{a,in} < 33.9\ ^\circ C$ $0.01 < d_{a,in} < 0.021$, kg/kg
Structured packing Cross-flow dehumidification $GaCl_2$	$\varepsilon_Y = \frac{1-\{0.363(G_{a,in}/G_{s,in})^{-0.038}\ \exp[1.012(T_{a,in}/T_{s,in})]/\pi^{0.342}\}}{1-\{0.267\ \exp[1.401(T_{a,in}/T_{s,in})]/\pi^{0.363}\}}$	Literature [14] $0.5 < G_s < 3.2\ kg/(m^2 s)$ $26 < T_{s,in} < 39\ ^\circ C$ $0.32 < X_{s,in} < 0.43$ $0.9 < G_a < 2.0$, $kg/(m^2\ s)$ $0.016 < d_{a,in} < 0.025$, g/kg $26 < T_{a,in} < 40\ ^\circ C$ Average error is $\pm 10\%$

packed tower. Based on the mass transfer coefficient from the Onda's model, a theoretical study was conducted to evaluate the heat and mass coefficients involving three liquid desiccants, namely calcium chloride, lithium chloride, and a mixture of 50% calcium chloride and 50% lithium chloride (called cost-effective liquid desiccant, CELD). The dehumidification performances of the three liquid desiccants were compared and discussed.

3.3. The overall convective model

In the overall convective model, the convective heat and mass transfer coefficients are the most critical parameters. The overall convective heat and mass transfer coefficients can be expressed by dimensionless parameters—*NTU* and *Le*. Here the *NTU* is the function of convective mass transfer coefficient, dimensions of the packing tower and airflow rate, and *Le* is the ratio of convective heat transfer coefficient and the product of convective mass transfer coefficient and specific heat of air. Therefore, the overall convective model is often called *NTU–Le* model.

Chung [20] adopted the Buckingham *Pi* method to analyze the convective heat and mass transfer coefficients between air and desiccant, and the correlations of the heat and mass transfer coefficients of the random and structured packing were obtained by using non-linearity regressive of experiment data in the dehumidification with three random packings and one structured packing. The experimental results showed that the error of the mass transfer coefficients of the structured packing was less than 10%. The error was mainly due to the effective mass transfer surface area.

Liu et al. [21–23] adopted the model to analyze the heat and mass transfer processes in a cross-flow dehumidifier/regenerator using liquid desiccant, and the *NTU* in the model was correlated of the experimental data of the dehumidifier and regenerator in structured packing. The similarity of coupled heat and mass between air–water and air–liquid desiccant direct-contact systems was disclosed to point out the reachable handling zone of the outlet air, which would guide the design of operation conditions of air and liquid desiccant and benefit the choices of dehumidification and regeneration parameters of air and desiccant.

Yin et al.[24] indicated that the overall convective heat and mass transfer often derived from experimental results and the determination of heat and mass transfer coefficients by the traditional log mean method would be not acceptable for the coupled heat and mass transfer. Considering the local distributions of temperature difference and humidity difference, they developed a new method called h_D–*Le* separative evaluation method for determining coupled heat and mass transfer coefficients between air and liquid desiccants, by which the heat and mass transfer coefficients between air and liquid desiccants were calculated from experimental inlet and outlet parameters of air and desiccant solution. The effects of the air and desiccant inlet parameters on the Lewis number, heat and mass transfer coefficients were discussed. The result showed that the Lewis number greatly depended on the operation parameters and conditions of the air and desiccant. This model can be solved by the numerical calculation methods, and its computational capacity is simpler than the double film model. It is especially applicable for the physical complicated flow, like the packed towers. The coupled heat and mass transfer coefficients support the basic data for the *NTU–Le* model and shape the exact prediction.

In recent years, the analytical solution of the *NTU–Le* model has been developed with some linear assumptions and simplifications. Ren et al. [25–29] utilized the *NTU–Le* model to establish the linear relationship between humidity/enthalpy efficiency and *NTU or Le* based on some simplifications and assumptions, such as the constant concentration of the solution and linear assumption of the relationship between air humidity and desiccant concentration. The analytical solution was obtained for the heat and mass transfer process of the packed bed and internally cooled/heated liquid desiccant–air contact units. Liu et al. [30–32] developed the analytical solutions of air enthalpy, desiccant equivalent enthalpy as well as enthalpy efficiency under several reasonable assumptions for cross-flow, counter-flow and parallel flow dehumidifier. The analytical solutions of enthalpy efficiency agreed well with the experimental findings, and the error was less than 20% under specific operation conditions. Davoud and Meysam [33] built the analytical model of the air dehumidification process with the assumption of the same equivalent moisture content of the desiccant under the high flow rate of the desiccant. The analytic solution provided the outlet parameters of the air and desiccant.

3.4. Performance evaluation of the liquid desiccant dehumidification

The moisture removal rate is often used for measuring the performance of handling the latent heat load of the processed air

$$\Delta d = d_{a,in} - d_{a,out} \tag{6}$$

The humidity efficiency is a dimensionless humidity ratio or vapor pressure ratio, which can give a preliminary prediction of the dehumidification performance. Dai and Zhang [6], Martin [9] and Gandhidasan [10] defined this in their studies by the following equation:

$$\varepsilon_Y = \frac{d_{a,in} - d_{a,out}}{d_{a,in} - d_{Ts,equ}} \tag{7}$$

where $d_{Ts,equ}$ is the humidity of the air in vapor pressure equilibrium with the desiccant solution, and is the assumed ideal humidity that the exit air could reach.

Enthalpy effectiveness ε_H is defined as the ratio of actual change in enthalpy of the air to the maximum possible change. Similar to humidity effectiveness, the maximum possible difference depends on the equilibrium with the inlet desiccant solution. Martin [9] defined it by the following equation:

$$\varepsilon_H = \frac{h_{a,in} - h_{a,out}}{h_{a,in} - h_{Ts,equ}} \tag{8}$$

where $h_{Ts,equ}$ is the enthalpy of the air in vapor pressure equilibrium with the desiccant solution, and is decided by the surface vapor pressure and temperature of the inlet desiccant solution.

The cooling performance of the air is also involved in the dehumidification process if the inlet air temperature is higher than the desiccant solution. Gandhidasan [10] defined the dimensionless temperature ratio by the following equation:

$$\varepsilon_T = \frac{T_{a,in} - T_{a,out}}{T_{a,in} - T_{s,in}} \tag{9}$$

Here, another efficiency of desiccant solution's concentration is given as follows [34]:

$$\varepsilon_c = \frac{c_{s,out} - c_{s,in}}{c_{s,sat} - c_{s,in}} \tag{10}$$

The desiccant solution's concentration at the saturation point at the average regeneration temperature is the ceiling line for the concentration of regenerated solution. $c_{s,sat}$ is the concentration of the saturated desiccant solution at the given average regeneration temperature, which should be in equilibrium with the inlet air. Exergy analysis is an effective way to analyze the air conditioning system, and it indicates the path for the system's thermodynamic performance improvement. Many researchers analyzed liquid desiccant air conditioning system by the second law of

thermodynamics, and especially focused on air exergy in the dehumidification and humidification. Assouad [35] carried out an exergy analysis of a solar powered liquid desiccant system using solar collector/regenerator. It was found that exergy was consumed mainly in the preheater, the regenerator and the absorber. Ahmed et al. [36] conducted exergy investigation on a hybrid system incorporating absorber and liquid desiccant dehumidifier, calculated the irreversible losses of the hybrid cycle and optimized the desiccant flow rate of the partly closed solar regenerator. Li [37] reported the changes of air exergy and designed the reversible dehumidification process between liquid and air, and the operating condition was analyzed for the reversible dehumidification process.

Dai [38] built a mathematical model for exergy analysis to investigate the performance of liquid desiccant dehumidification and cooling system based on LiCl solution. The effort focused on exergy loss in each component as well as the influence of desiccant solution and air inlet parameters on the dehumidifier (demonstrated by exergy efficiency of the dehumidifier). The results showed that the greatest part of exergy loss occured in solution-hot water heat exchanger, 24.5%, followed by solution–solution recuperator and cooling water–solution heat exchanger, accounted for 24.4% and 22.8% respectively. The performance of the system can be expressed by the exergy efficiency, and is defined as the ratio between the exergy increase of the treated air and the total exergy of the whole system:

$$\varepsilon_H = \frac{m_a(e_{a,out} - e_{a,in})}{m_{hot}(e_{hot,in} - e_{hot,out}) + m_{cool}(e_{cool,in} - e_{cool,out})} \tag{11}$$

The exergy efficiency can be used to evaluate the performance of liquid desiccant dehumidification, given as.

$$\varepsilon_H = \frac{m_a(e_{a,out} - e_{a,in})}{m_{s,in}e_{s,in} - m_{s,out}e_{s,out}} \tag{12}$$

Dai [39] developed a novel two-stage liquid desiccant dehumidification system assisted by calcium chloride ($CaCl_2$) solution, and optimized the thermal coefficient of performance through exergy analysis based on the second law of thermodynamics. The exergy loss in the desiccant–desiccant heat recovery process can be significantly reduced by increasing the desiccant concentration variance between the strong desiccant solution after regeneration and the weak desiccant solution after dehumidification. Meanwhile, the pre-dehumidification of $CaCl_2$ solution can reduce the irreversibility in the regeneration/dehumidification process. Compared to the basic system, the thermal coefficient performance and exergy efficiency of the proposed system are increased from 0.24 to 0.73 and from 6.8% to 23.0%, respectively, under the given conditions.

Kanoglu [40,41] developed the energy and exergy analyses of open-cycle desiccant cooling systems, and the exergy destruction and exergy efficiency relations were derived for the system and its components. The relations are applied to the experimental units using the data collected during a typical operation. The experimental system has a COP of 0.35, a reversible COP of 3.11, and an exergy efficiency of 11.1%. Desiccant wheel has the highest percentage of total exergy destruction with 33.8%, followed by the heating system with 31.2%.

The thermal efficiency of the regeneration process can be calculated using two models [42]. In one model heating the solution drives the regeneration process, and in the other model heating the air drives the regeneration process. Different models have different thermal efficiencies. The thermal efficiency of regeneration process of the first model is given by Eq. (13), and the other one is given by Eq. (14):

$$\eta = \frac{M_a r(d_{a,out} - d_{a,in})}{M_s Cp_s(T_{s,in} - T_{s,o})} \tag{13}$$

$$\eta = \frac{r(d_{a,out} - d_{a,in})}{Cp_a(T_{a,in} - T_{a,r})} \tag{14}$$

$T_{a,r}$ is the reference temperature of the air, and is usually 25 °C. The above-mentioned four dimensionless ratios, coupled with the related energy balance equations can be used to predict the mass removal rate of the moisture with the known initial conditions of air and desiccant solution.

4. Development of dehumidifiers and regenerators

The dehumidifier and regenerator are the most important components in liquid desiccant air conditioning systems (LDACs). The vapor pressure difference between the air and liquid desiccants is the driving force of the mass transfer process. The regeneration of liquid desiccant can be driven by low-grade heat, such as solar energy, waste heat or other low-grade heat sources. The heat and mass transfer performances in the dehumidifier/regenerator greatly influence the performance of the LDACS. The spray tower, wetted wall tower and packed tower are often used as the equipment for heat and mass transfer between air and liquid desiccants. The advantages of spray tower are that the air side pressure drop is very small, and it can also offer a large gas–liquid contacting area for high viscosity liquids; meanwhile, spray tower can be clogged by the dirt and has some potential drawbacks, such as a large number of droplets are carried into the air. Wetted wall tower can offer a stable-flow condition and a constant gas–liquid contacting area for gas and liquid, but the surface area is relatively small. Packed towers can provide a large gas–liquid contacting area with acceptable air pressure drop. For these situations, many researchers often adopted packed towers as the dehumidifier and regenerator. Random packing and structured packing are often used in packed towers.

The random packing is relatively cheap and has a large surface area, but the air side pressure drop is relatively large. Many researchers conducted some experimental studies on the dehumidifier/regenerator using random packing towers. Random and structured packings were compared for their efficiency in dehumidification of air in a packed column using lithium chloride solutions by Chung and Ghosh [20]. Experiments were conducted with cross-corrugated cellulose and poly(vinyl chloride) (PVC) structured packing, and the results were compared with the data previously obtained for polypropylene Flexi rings and ceramic Berl saddles as random packing. The data obtained with 5/8 in. polypropylene Flexi rings, 1/2 in. ceramic Berl saddles, and 2 in. clay Raschig rings were used to test for random packing. Potnis and Lenz [43] adopted lithium bromide as the liquid desiccant to investigate the random and structured packing towers with varying bed depths in the regenerator and the dehumidifier of a solar-assisted liquid-desiccant system. The results indicated that the condition for the liquid phase was turbulent for the operating conditions in both contactors. Liquid-phase mass-transfer coefficients for the packed bed were obtained. The random packing mass transfer coefficients varied from 0.48 to 2 mol/(m^2 s), while the double-layer, structured packing mass-transfer coefficients varied from 0.018 to 0.035 mol/(m^2 s). The mass-transfer coefficients were converted into a dimensionless form based on diffusivity values obtained experimentally. Longo and Gasparella [44] performed the experimental tests on the desiccant regeneration in a packed column with lithium bromide. The tests showed that

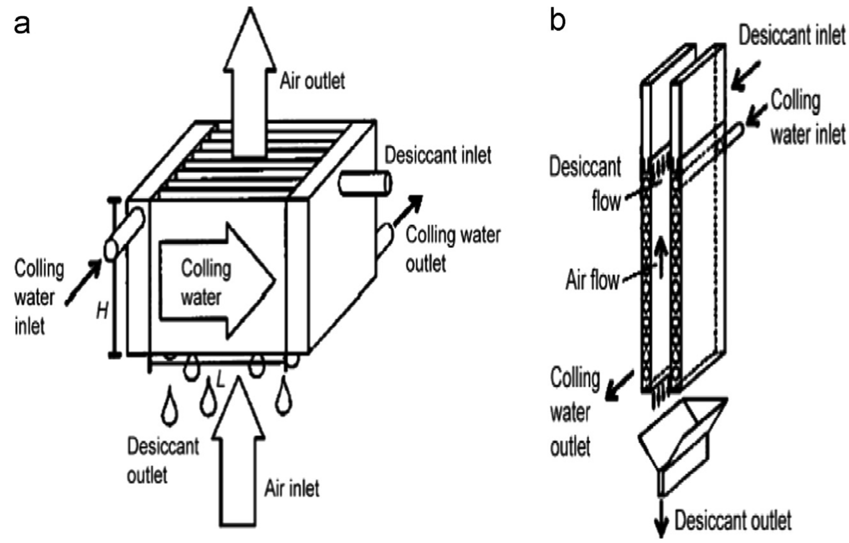

Fig. 3. A water-cooling flat dehumidifier.

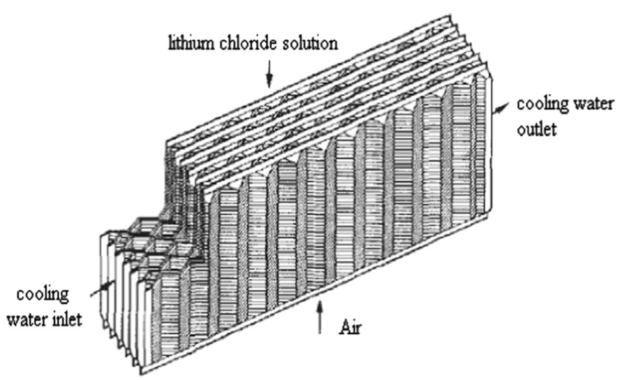

Fig. 4. Internally cooled dehumidifier with corrugated plates.

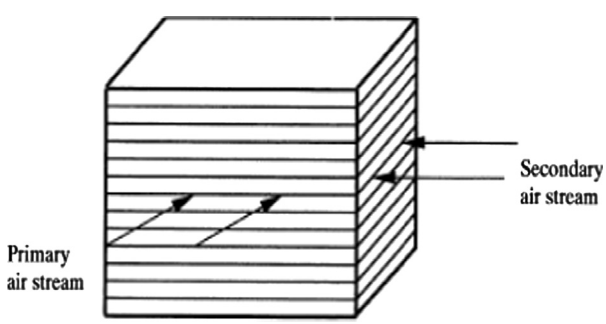

Fig. 5. Internally cooled cross-flow dehumidifier.

desiccant regeneration required the temperature level to be around 50 °C, which could be easily obtained by solar energy or industrial waste heat. The regeneration performance of the random column showed 20–25% higher than structured column, whereas the structured column showed air side pressure drop 65–75% lower. Lazzarin [45] used lithium bromide as the liquid desiccant to study experimentally the counter-flow packed tower dehumidifier and explored the effects of gas–liquid mass flux ratio on dehumidify efficiency and air moisture removal (shown in Fig. 2).

Structured packing is trimly arrayed in packed tower, and provides uniform gas–liquid flow path. Air side pressure drop is relatively small and can offer a large surface area. Therefore, in recent years the structured packed tower has widely been used in various kinds of the parallel-flow, counter-flow or cross-flow dehumidifiers. Depending on whether cooling or heating exist in the regenerators and dehumidifiers, it can be divided into adiabatic type and internally cooled (heated) type. The adiabatic dehumidifiers can be found in a wide range of industrial and residential applications, and can afford a large contacting area and high mass and heat transfer efficiency. However, it has the potential drawback of the significant increase in the liquid desiccant temperature owing to the latent heat release during the moisture removal process, which can cause the dehumidification decay. For these situations, the internally cooled dehumidifier using cooling coils to remove the heat generated from dehumidification is a sound alternative. The cold and heat sources can adopt water or refrigerants combined with heat pump systems.

Fig. 3 shows a water-cooling flat-plate dehumidifier [46,47]. Cooling water circulated in polypropylene double plates involved an upward airflow and a downward solution flow. A special distributor of liquid desiccant on top of each plate could help the solution flow uniformly over the exchanger surface. Fig. 4 shows an internally cooled dehumidifier consisting of a corrugated plate [48]. Compared with the flat-plate dehumidifier, it has a higher dehumidification performance due to the air turbulence and better water-cooling performance. Fig. 5 shows a flat dehumidifier cooled by water evaporation [49,50]. The liquid desiccant and cooling fluid were with a cross-flow configuration that accompanied water evaporative cooling in the internally cooling channels.

Jiang et al. [51,52] proposed a multi-stage dehumidifier with an auxiliary cooling module, which could cool the desiccant solution, as shown in Figs. 6 and 7. In the system shown in Fig. 6, the desiccant solution was cooled by the external cold source, and the dehumidification performance could be improved. In the second system shown in Fig. 7, the high temperature heat load can be taken away by the normal cooling source with the temperature of 26–30 °C, and the cold water with the temperature of 18–21 °C can cool the desiccant solution in the low moisture content part, and the benefit was reducing the load of the cold water (18–21 °C). Guo et al. [53] developed the fin-plastic tube heat exchanger for liquid desiccant dehumidifiers and regenerators with excellent corrosion prevention performance. It could be used for the internally cooled/heated dehumidifier/regenerator in liquid desiccant system (shown in Fig. 8).

Yin et al. [54,55] developed a new type of internally cooled/heated dehumidifier/regenerator based on the plate-fin heat exchanger (PFHE), as shown in Fig. 9. The desiccant solution was

distributed to the fins and flowed down by the gravity, and the air was blown from the bottom with a counter-flowing configuration. It could control the water temperature to realize the process of liquid desiccant dehumidification/regeneration. When the water temperature was high, the hot water was offered to realize desiccant regeneration. On the contrary, the cooled water realized the dehumidification of the air. The experimental results showed that the internally cooled dehumidification and internally heated regeneration performance were significantly better than that of the adiabatic one; moreover, the internally heated regenerator could offer better thermal effectiveness, which means that internally heated regenerators can provide higher energy efficiency. The results indicated that internally heated regenerators were very suitable for the conditions of low flow rate of liquid desiccant. Therefore, it is promising a alternative to develop zero-carrier regenerators.

5. Solar energy utilization in desiccant cooling

In the liquid desiccant cooling systems, liquid desiccants are often heated in regenerators and gets concentrated. Solar collectors were assumed to be used for regeneration in the liquid desiccant cooling system. A solar energy driven dehumidification system was first carried out in the USSR by Kakabaev et al. [56] in 1969. Solar collectors can be used directly and indirectly for desiccant regeneration. Solar collection and liquid desiccant regeneration can be set up separately, called indirect solar desiccant regeneration [57]. Many researchers proposed combining solar collection with desiccant regeneration together to develop the solar desiccant collector/regenerator, which was called direct solar desiccant regeneration. Haim [58] simulated the two open-cycle absorption refrigeration systems with direct solar desiccant regeneration and indirect desiccant regeneration, and found that the performance of direct solar desiccant regeneration was better than the indirect solar desiccant regeneration.

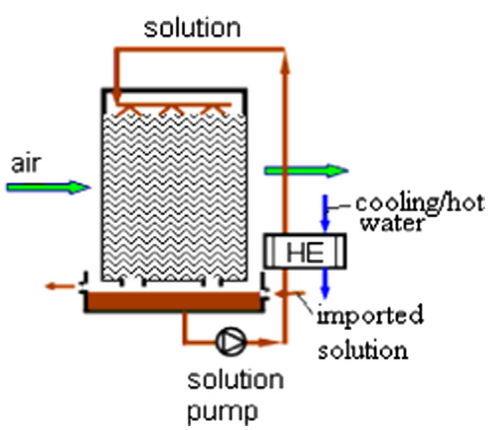

Fig. 6. Dehumidifier/regenerator with auxiliary cooling/heating module.

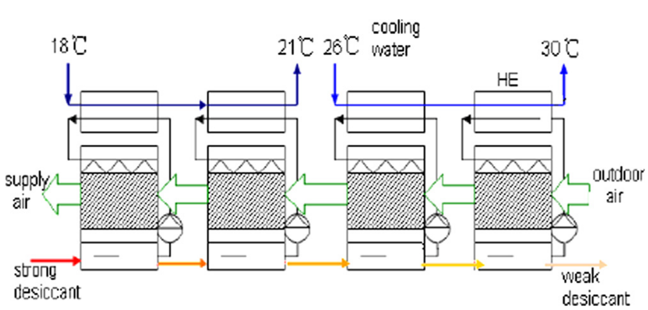

Fig. 7. Multi-stage dehumidification.

Fig. 10 shows a new type of solar liquid desiccant cooling system with a solar *C*/*R* [59], includes three core components—solar collector/regenerator(*C*/*R*), air dehumidifier and evaporative cooler. The working fluids ran in two loops: air loop and liquid desiccant loop. The processed air was dehumidified in the dehumidifier, and then cooled by cooling water. The diluted solution leaving the dehumidifier was heated in the heat exchanger, and then entered into the solar collector/regenerator to be heated further to the regeneration temperature and regenerated. The feasibility and performance of the direct solar desiccant regeneration through the *C*/*R* component were analyzed.

5.1. Solar collector regenerator with natural convection

Earlier the natural convection was adopted to realize the solar collection/desiccant regeneration on pitched roofs [60]. Nelson et al. [61,62] designed a natural convection glazed collector/regenerator and explored the evaporation rate model to conduct a comparative study with an unglazed one. Gandhidasan et al. [63–65] proposed a partly closed solar collector/regenerator, as shown in Fig. 11. The solar collector/regenerator was divided into glass-covered section and an open section. In the glass-covered section, the solution was heated to the required regeneration temperature. In the open section, the solution was indirectly in contact with the air for the heat and mass transfer to realize the liquid regeneration. Kakabaev et al. [66] found that the mass transfer coefficient and exit solution temperature increased linearly with solar radiation. McCormick et al. [67] reported that if the height of the glazing exceeded a certain value, its performance competed with that of unglazed solar collector regenerator.

5.2. Solar collector regenerator with forced convection

Yang et al. [68–72] conducted much theoretical and experimental research on the forced convection solar collector/regenerator (shown in Fig. 12) and found that the *C*/*R* efficiency of the counterflow was better than the parallel flow. They designed a double glass-covered collector/regenerator as shown in Fig. 13.

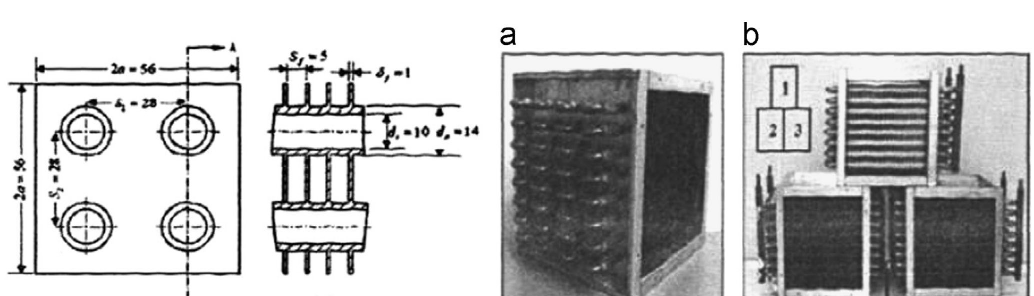

Fig. 8. Structure drawing and physical drawing of the fin-plastic tube heat exchanger.

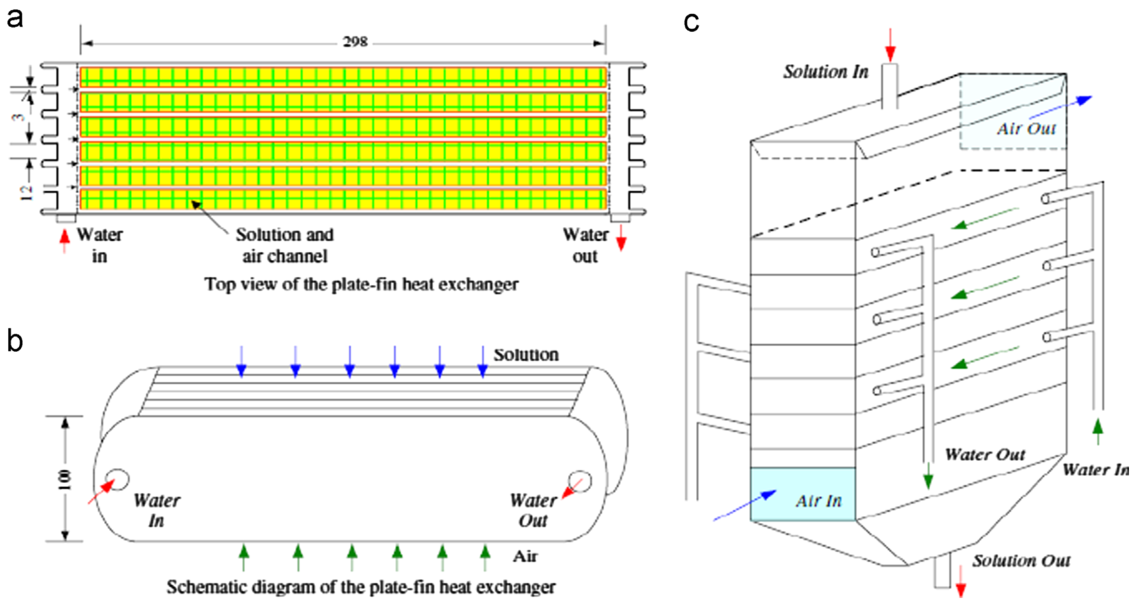

Fig. 9. Schematic diagram of a new type of internally cooled/heated dehumidifier/regenerator.

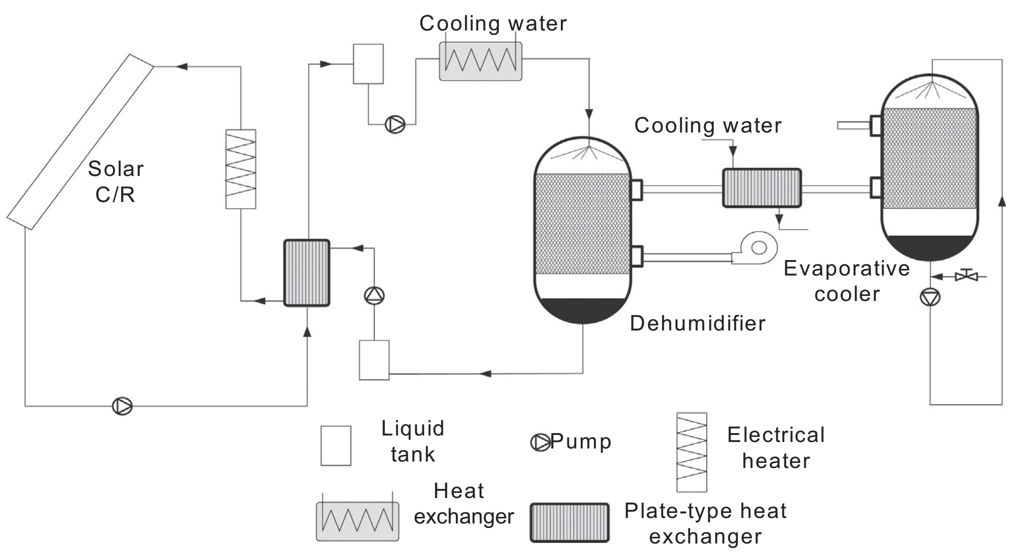

Fig. 10. Flow chart of a solar liquid desiccant cooling system.

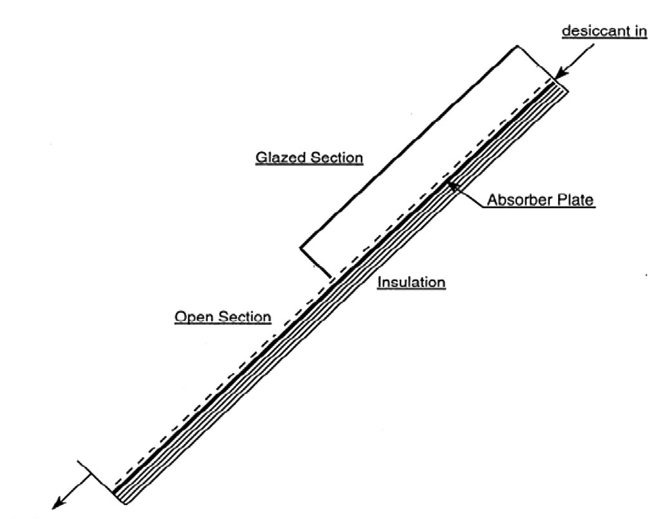

Fig. 11. Schematic of a partly closed-open solar collector/regenerator.

The double-glazed *C/R* with forced convection indicated a better performance than the natural convection single-glazed *C/R*.

Alizadeh et al. [73,74] carried out an experimental study on the forced parallel-flow solar collector/regenerator and found the optimum air to solution mass flow rate for the maximum water evaporation rate under the given operating conditions. Fig. 14 shows the schematic diagram of the parallel-flow collector/regenerator, and the results indicated that the evaporation rate increased with the airflow, and decreased with the solution flow; and the overall performance greatly depended on the solar radiation.

Kabeel [75] performed a comparative study on the natural convection with the forced-convection solar regeneration (shown in Fig. 15). The experimental correlations of the regeneration efficiency were obtained. It was found that the forced convection cross-flow solar regenerator was with higher efficiency and the mass transfer coefficient of the forced unit was much higher than the free one.

Liquid desiccant regeneration performance is often relatively low in hot and humid climate. To improve the regeneration

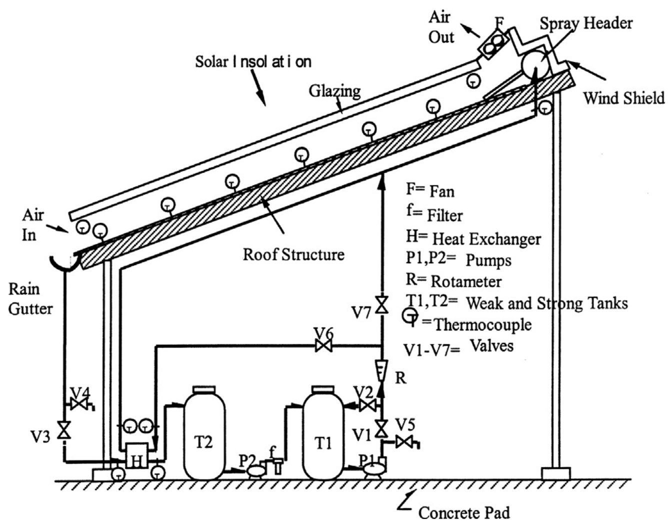

Fig. 12. Schematic of the experimental *C/R*.

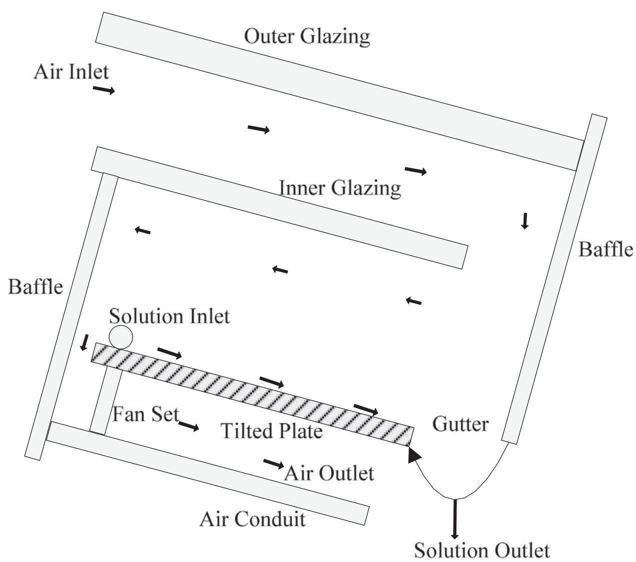

Fig. 13. The double glass covered *C/R*.

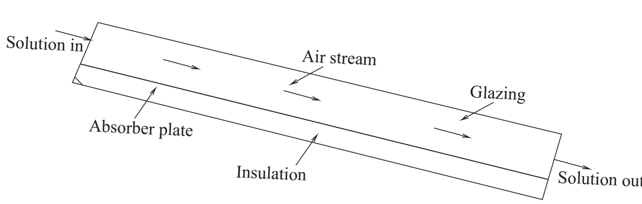

Fig. 14. Schematic of the parallel flow solar regenerator.

performance, Peng et al. [76,77] proposed a new type of solar liquid collector/regenerator as shown in Fig. 16. There was an air pretreatment unit for less humidity ratio of the regeneration air in the new regeneration system. Part of the concentrated solution out of liquid heat exchanger at low temperature was delivered into air pretreatment and dehumidified the regeneration air; then the desiccant entered the collector/regenerator via a liquid–liquid heat exchanger. The other part of the concentrated desiccant went for dehumidification of the processed air. The diluted solution from the dehumidification system came back to the collector/regenerator together with the desiccant from the pretreatment unit. For air circulation, the outdoor hot and moist air was dehumidified in the air pretreatment unit and then entered into the solar collector/regenerator where the air was heated and humidified, and then was vented into the atmosphere. The solution outlet concentration of the *C/R* in solar air pretreatment collector/regenerator system would be higher than that in the traditional *C/R*. Compared with the traditional *C/R* system, the outstanding advantage of the air pretreatment collector/regenerator was that it could utilize lower grade heat to achieve better regeneration performance, particularly suitable for the regions with high humidity.

5.3. *New application of liquid desiccant dehumidification*

In recent years, many experts combined the traditional vapor compression refrigeration systems with liquid desiccant dehumidification to develop new air conditioning systems, which were temperature and humidity independent control air conditioning systems. The new air conditioning system could operate at high COP with a high evaporation temperature for sensible cooling.

Yin et al. [78] presented a novel liquid desiccant air conditioning system, the liquid desiccant evaporation cooling air conditioning system (LDCS), and is shown in Fig. 17. In this system, it included a dehumidifier, a regenerator and an evaporative cooler. Both air and concentrated desiccant film with low temperature entered the dehumidifier, and the water vapor was transferred from the air to the desiccant film. In the regenerator, the diluted desiccant solution from the dehumidifier was heated to a higher temperature by low-grade heat source and the water vapor was transferred from the desiccant solution to the air. The dehumidified air went into the evaporative cooler to get cooled and humidified, and then was sent to the air conditioning space. The system could be driven by low-grade heat sources, such as solar energy and industrial waste heat with the temperatures between 60 °C and 80 °C. Yin et al. [79] proposed a novel liquid desiccant air conditioning system and is shown in Fig. 18. The new system consisted of four units: liquid desiccant (LiCl–H_2O) regeneration unit, air dehumidification unit, producing chilled water unit by evaporative cooling, and radiant cooling and dehumidified ventilation unit. The liquid desiccant regeneration unit was used to concentrate the diluted liquid desiccant from the dehumidifier by solar energy. The air dehumidification unit used liquid desiccant to produce very dry air for evaporative cooling unit and to condition the space. The air conditioned space could be kept as a thermally comfortable environment by radiant cooling and ventilation with dry air. The dry air was a part of the dehumidifier. The other part of the dry air entered the evaporative cooler to produce chilled water for the radiant ceiling panels. The temperature of the chilled water could be as low as 12 °C with good thermal performance. The dehumidifier in the system was different from the one in Fig. 17. The humidity ratio of the air leaving the dehumidifier in Fig. 17 should be much lower than that shown in Fig. 18.

Lafuenti et al. [80] proposed some system innovations in solar desiccant cooling and absorption chiller, aiming at higher energy-efficiency technology for hot and humid climates. Four air treatment units were analyzed using suitable mathematical models, and the results showed that the most efficient solution in terms of COP_{UTA} was the desiccant cooling system with partial recirculation of indoor air. Halliday et al. [81] and Mavroudaki et al. [82] conducted feasibility study of solar driven desiccant cooling in diverse European cities representing different climatic zones. The results revealed the primary energy savings potential in all climatic conditions. A decline in energy saving was noticed under highly humid zones. The decline was attributed to the high temperature required to regenerate the desiccant in the climates of high humidity.

Henning et al. [83] conducted a parametric study on a combined desiccant/chiller solar assisted cooling system and showed

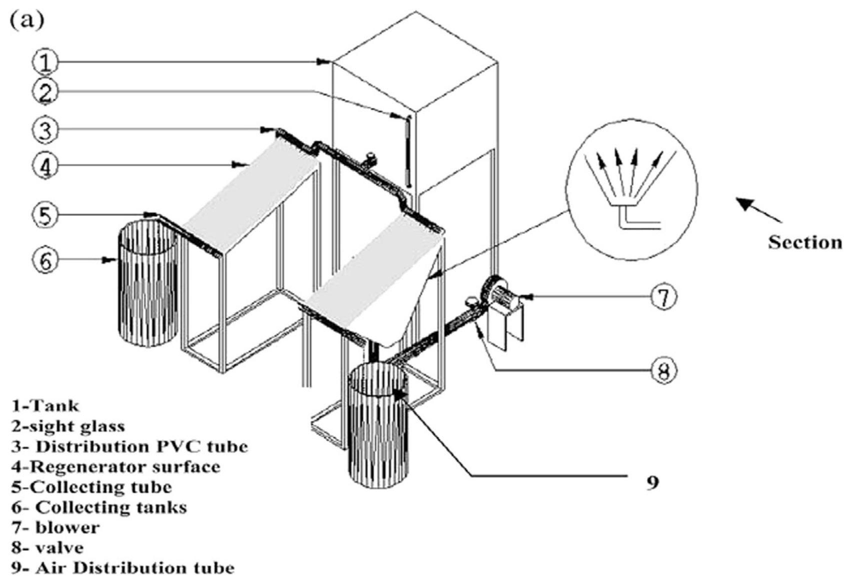

Fig. 15. Layout of the proposed system.

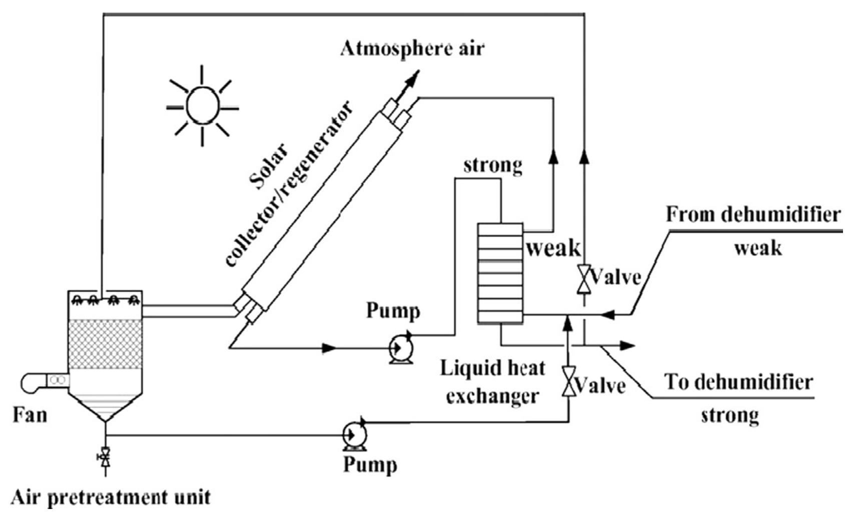

Fig. 16. Schematic diagram of solar air pretreatment *C*/*R*.

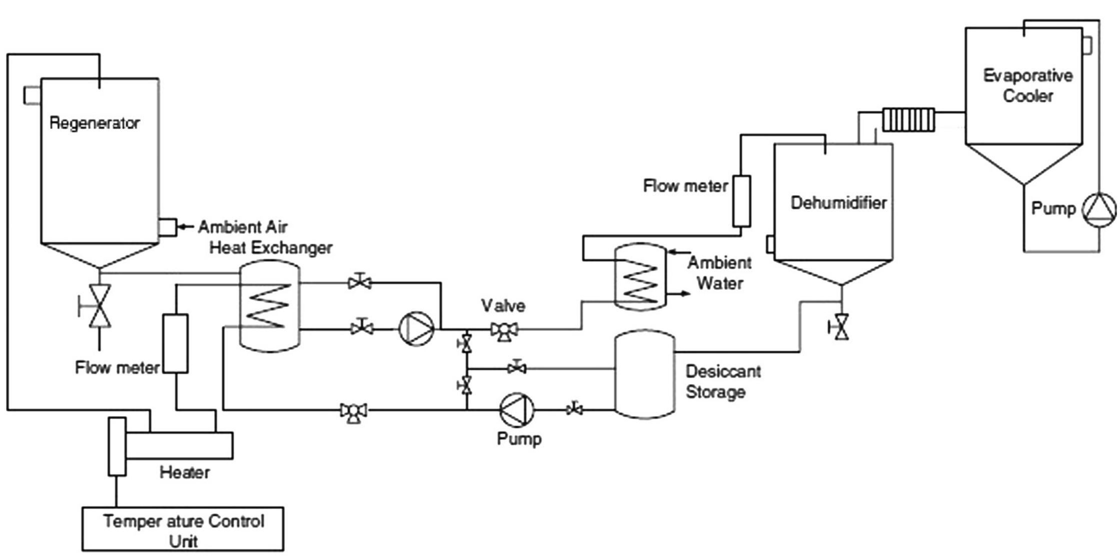

Fig. 17. Schematic diagram of the liquid desiccant evaporation cooling air conditioning system.

not only the feasibility but also the primary energy savings of up to 50% with a low increase of overall costs. Jiang et al. [84,85] presented a heat pump driven two-stage liquid desiccant fresh air unit; and the full load COP of the fresh air unit was 5.0, and even over 5.9 under partial load conditions. Besides, the full load COP of the integrated heat pump was 4.01, and the part load COP could be as high as 5.72. The sensible heat recovery efficiency of the solution heat exchanger was over 80%. Davies [86,87] used the liquid desiccant dehumidification with solar regeneration to lower the temperature in greenhouses. Compared to conventional evaporative cooling, the desiccant system lowered the average daily maximum temperatures in the hot season by 5.5–7.5 °C, sufficient to maintain viable growing conditions for lettuce throughout the year.

Lowenstein et al. [88,89] investigated the problem of the solar liquid desiccant dehumidification air conditioning in desiccant droplets carrier, and the results showed that it was decreased when the flow rate was low. Lazzarin et al. [90] used a liquid

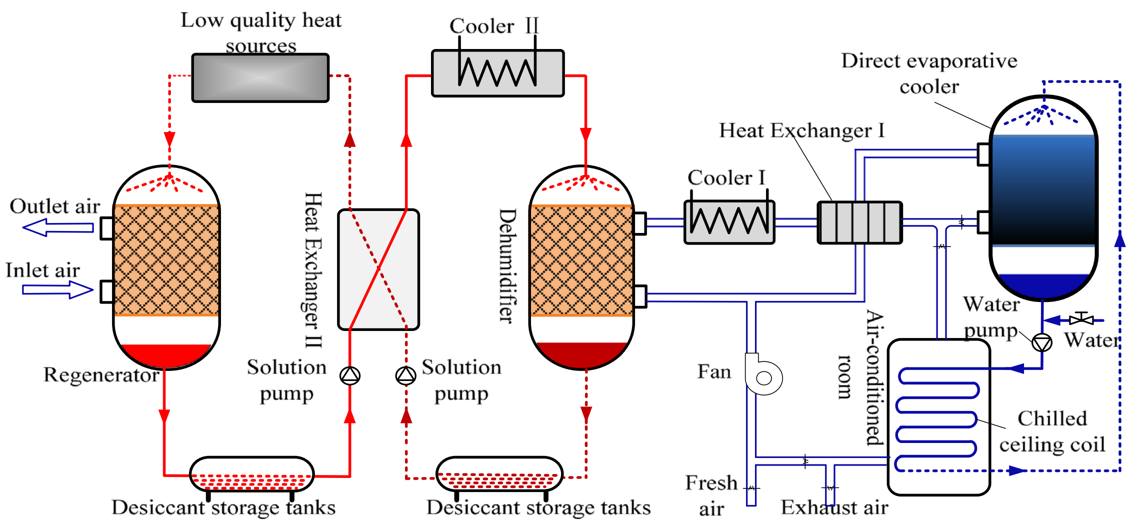

Fig. 18. Schematic diagram of the direct evaporative cooling air conditioning system based on liquid desiccant.

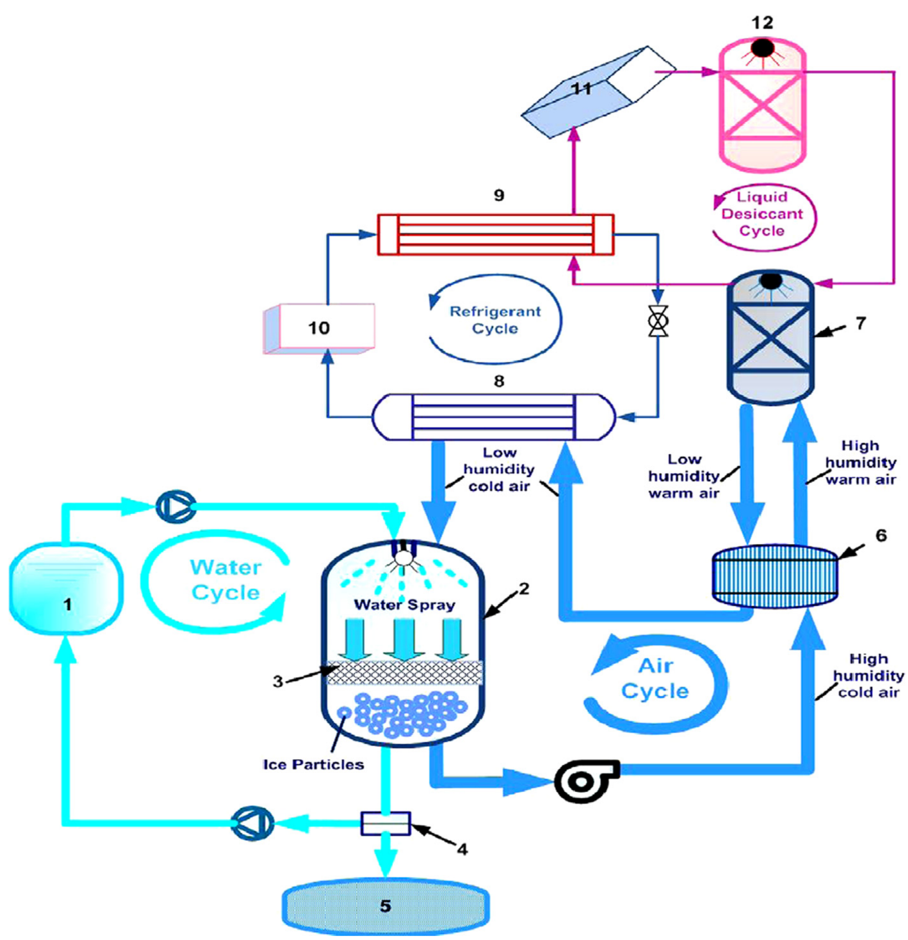

Fig. 19. Schematic diagram of the novel ice slurry production system.

desiccant system in different seasons for the air conditioning in a university building. In summer, the plant dehumidified the ambient air to remove the latent load, and in winter the system operated an effective heat recovery of the exhausted air. Li et al. [91–94] proposed a novel ice slurry production system to improve the conventional ice-slurry producing method by deep dehumidification and evaporative cooling, which consisted of two major parts: the liquid dehumidification process for very dry air and the evaporative super-cooling thorough the very dry air, shown in Fig. 19. Water could be cooled down through its direct evaporation to air and the final water temperature depended on the air wet-bulb temperature. When the air was very dry, the temperature of water would drop. If the vapor pressure of the atmosphere was below 611 Pa (the vapor pressure is around 611 Pa at 0 °C), the water would keep on evaporating until the vapor pressure balance was achieved between the water and the air, accompanied with the water temperature falling below 0 °C. Water could be cooled to 0 °C when the vapor pressure in the air is less than 611 Pa. Therefore, water drops could be super-cooled by water evaporation and then turn to ice particle. The very dry air could be obtained through the desiccant dehumidification. It is a novel ice slurry making method combining desiccant dehumidification and evaporative cooling, and can avoid the ice block risk during ice making using the traditional super-cooling method. It is easy for the novel ice making method to combine with vapor compression refrigeration system. The refrigeration system can not only pre-cool the water or air, but also contributes to the normally wasted heat from its condenser for the desiccant dehumidification system. That is an important improvement because the double effects alleviate the burden of electric power and the whole performance is improved. Besides, the mechanism of the phase change from super-cooled water to ice needs more research work. Also, more attention should be paid to the sustainability and reliability of this new ice slurry producing method.

6. Comparative analysis of the energy savings of the liquid desiccant air conditioning system against traditional air conditioning system

The traditional vapor-compression refrigeration and air-conditioning systems are consuming plenty of electricity. Much work has been carried out to develop energy saving technologies to increase the energy efficiency. Air conditioning technology based on liquid desiccant dehumidification is one of them. Dai et al. [95,96] made a performance analysis on a hybrid air-conditioning system, which consisted of desiccant dehumidification, evaporative cooling and vapor compression air conditioning. Experimental results showed that the hybrid cooling system produced more cooling capacity than the VCS alone by 20∼30%.

Grossman [97] carried out theoretical analysis on a low-grade heat driven open desiccant cooling air conditioning system and the results indicated that the thermal performance enhanced with the increase of heat source temperature and the maximum thermal *COP* was around 0.45. The thermal *COP* of the system could reach around 0.8 when only used for dehumidification at Haifa in Israel [98].

Tang and Liu [99] compared energy consumptions of the traditional air conditioning system and liquid desiccant air conditioning system in industrial buildings in Shenzhen. The results indicated that under the outdoor design condition in summer, the COP of the traditional air conditioning system was 2.94, while that of the liquid desiccant air conditioning system was 5.42. All the results verified the advantage of liquid desiccant air conditioning systems in energy saving.

7. Conclusions

A detailed state-of-the-art review of the heat and mass transfer models between air and desiccant was pursued in this report. Much effort was focused on the models of heat and mass transfer between air and desiccants. Three models that were usually used indicated very exclusive privileges in special cases. The finite control volume model is very suitable for theoretical analysis with precise prediction and acceptable calculation inputs. Recent advances in the model development focused on the highlight in the *NTU–Le* model, and its application in description of complex heat and mass transfer processes. The determination of heat and mass transfer coefficients is very contentious due to the complications presented by the contacting material, operation conditions and types of desiccants, and the resolution from the experimental data. It also has a direct impact on the precision of the *NTU–Le* model, and therefore further investigation is necessary to clarify it.

Packing towers are leading the liquid desiccant dehumidifiers and regenerators owing to the significant specific area for heat and mass transfer. Internally heated regenerators have been emerging in recent years and they were verified to offer better thermal efficiency and fewer carriers of liquid drops. Solar energy is very suitable for the liquid desiccant cooling system, and solar collector/regenerator are the typical components in liquid desiccant regeneration. By contrast, effective solar regenerators should be developed and improved.

Liquid desiccant dehumidification technology has been shown to be very effective in removing the moisture from air using low grade heat. Using the technology, energy-efficient air conditioning systems have been developed, which demonstrated superiority over the traditional vapor compression type system by allowing both temperature and humidity being controlled independently, and this is especially relevant in the regions with high air humidity. It is very attractive as an energy-efficient alternative air-conditioning system.

The new applications of liquid desiccant dehumidification technology were also discussed. It can be used in new air-conditioning systems, such as the liquid desiccant evaporation cooling air conditioning system, fresh air handling unit using liquid desiccant driven by a heat pump. A new ice slurry production system based on the liquid desiccant was proposed, and this could be applied in industrial areas like mass food storage and energy storage.

Acknowledgments

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